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EXPERIMENTAL INVESTIGATION OF TURBULENT FLOW IN TRANSITION DUCT

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ABSTRACT

An experimental study was conducted to investigate turbulent flow through from rectangular cross-sectional area to rectangular transition duct. Three kinds of channels with different length, side angle and geometrical dimensions were designed and then flow characteristic of them were all tested. The transition duct have different inlet and outlet cross-sectional areas, outlet section is double than inlet sectional area. Measurements were made at several station along the channel and at various flow-rates, Reynolds numbers ranging from 2.10^5 to 6.10^5, in which the two–dimensional mean, local velocity and pressure were measured at the x/l = 0.20, 0.40, 0.60, 0.80, 1.00 ratio, inlet by using hot wire anemometer. Based on experimental results, different flow characteristics were obtained. According to mean and local velocity and pressure drop values, each section point of the transition duct showed various values. Friction coefficient was lowered with increasing pipe length and increasing Reynolds number. Cross flows was shown in the short pipe along length at the low Reynolds.

Keywords: Transition duct, complex turbulent flow, secondary flow, flow measurement

1. Introduction

Pipes having same or different cross-sectional are used for transition ducts which are known as "transition pipes" or "transition fittings". This type of pipes can be seen at current machines and pipe line network. For instance transition parts of air condition channels, water and wind tunnels entrance regions of current machines pump and ventilator networks. Ducts with non-circular cross sections are also frequently encountered in industrial heat transfer equipment, compact heat exchanger, cooling channels in gas turbine blades, nuclear reactors, ventilation and turbo machinery, etc. The main flow in such ducts is influenced by the secondary motions in the plane perpendicular to the stream wise direction, commonly referred to as secondary flows of Prandtl's second-kind. The secondary motions may distort the axial flow and induce a reduction of the volumetric flow rate due to a considerably friction loss, especially in corrugated ducts. These motions are of major concern since they redistribute the turbulence kinetic energy in the cross section of the duct [1], which in turn affects the heat flux and temperature field distributions.

In general, power plants and micro turbine systems are designed to obtain high effectiveness and low pressure losses minimum volume and weight, and high reliability and low cost [2]. Applications, generally the heat exchangers contain flow channels with various cross-sectional shapes, curved, expanded-narrow channel or wavy in the main flow direction, to enhance the heat transfer.

Over the two decades, constant-extensional rate or transitions channel systems have become a rapidly developing technology, finding applications in many areas of engineering and science. Transition channel involve the flow of liquid or gas to accomplish their design purpose. While many transition channels including such micro scale analysis system as gas chromatographs and blood , are characterized by low Reynolds number flow, there are some application areas where the Reynolds number can be much higher, such as the development of micro scale heat exchangers of heat sinking applications, such as microelectronics cooling [3].

Wu at al. [4, 5] performed some of the first experiments in turbulent micro channel flow, measuring the friction factors and heat transfer characteristic of gas flow in etched glass and silicon micro channel. Their result showed that the friction factors and Nusselt number for both the laminar and turbulent flow regimes were larger than predictions using traditional macro scale correlations. Flow in small ducts has been studied by a many researches over the years. Schlichting [6] documents theories and experimental data from the pioneering works by Poiseuille et al. [7].

Pipes having different section geometry and cross-sectional area, there is enough information about for flow, but in most applications section geometry and cross-sectional area not constant. That kind of pipes transition systems needs a joining-part which has variable section geometry and cross-sectional area. However, there are few works on turbulent flow in transition pipe with a rectangular section, which is important for practical goal.

The purpose of this paper is to provide detailed information in the turbulent flow through a rectangular - sectioned to rectangular section.

2. Experimental

Experimental set up is given in Fig.1, in which consist of velocity measurements hot wire anemometer, a blower, a rotameter to measure the flow rate, and by-pass arrangement to control the flow. The test fluids were driven by blower. Air at the room temperature was supplied to the straight test duct through a settling chamber. Flow was set by a regulator and the flow rate was controlled by valve downstream of the test section. Flow strengtheners and mesh screens make the approaching flow uniform and parallel. Test fluid, after a large section, air, passes from constant section of air tunnel (300x300) enters to adaptor than goes to 1500 mm length of flow development channel which has a dimension of (140 x 200 mm). Transition test ducts have a rectangular section cross-section of entry 140 mm x 200 mm and exit 200 mm x 280 total length of the duct; $L_1 = 500 mm, L_2 = 750 mm\, and\, L_3 = 1000 mm$. Transition duct have; A_1 and A_2 show the entry and exit cross-sectional areas of the rectangular duct.

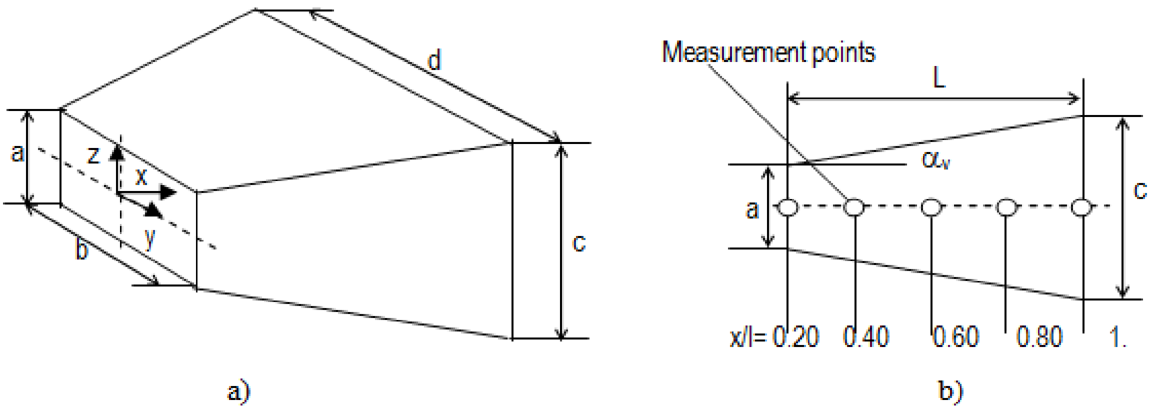

Fig. 1 – Experimental set up, a) Perspective projection b) Measurement holes

Notation

A - area, m^2

D_h - hydraulic diameter of the channel, m

f - friction factor

g - gravitational acceleration, m/s^2

L - length of the tube, m

Q - volume flow rate of water through the tube, m^3/s

Re - Reynolds number $(u_{av} D/\upsilon)$

U - velocity component in flow direction, m/s

u_{av} - the average velocity, m/s

x - coordinate in the axial/flow direction

ΔP - pressure gradient across the tube, N/m^2

Greek symbols

μ - dynamic viscosity of air, kg/msn

υ - kinematics viscosity of air, m^2/s

ρ - density of air, kg/m^3

$\alpha_{v,k}$ - α_v angle between top and bottom axes, α_k angle between side edges

ϕ - conical angle, $\circ$

α - a/b

β - c/d

Subscript

a,b - the dimensions of entry duct

c,d - the dimensions of the exit duct

v - vertical direction measurement

a - axial direction measurement

The distance, x, from the entry of Ax area, is not dependent upon the entry and exit cross-sectional area.

$$\bar{A}x = \frac{Ax}{A_1} = 1 + f_1(\alpha,\beta,K).\bar{x} + f2(\alpha,\beta,K).x^2 \qquad (1)$$

Here:

$$K = A_2 / A_1 \qquad (2)$$

And a-b is the dimensions of entry and c, d is the dimensions of the exit.

$$\alpha = {}^{a}\!/\!_{b}; \qquad 0 \le \alpha \le 1 \qquad (3)$$

$$\beta = {}^{c}\!/\!_{d}; \qquad 0 \le \beta \le 1 \tag{4}$$

Hence l is the length of the transition duct.

$$\bar{x} = \frac{x}{L} \tag{5}$$

The function of $f_1(\alpha,\beta,K)$ were found as shown in [9] and [10] $f_2(\alpha,\beta,K)$ is calculated as follows:

$$f_2(\alpha,\beta,K) = K - 1 - f_1(\alpha,\beta,K) \tag{6}$$

Here, ϕ , is conical angle of the rectangular duct, is calculated as follow

$$A_1 = \pi . R_1^2 \quad ; \quad A_2 = \pi . R_2^2 \tag{7}$$

$$R_1 = \sqrt{\frac{A_1}{\pi}} \quad ; \quad R_2 = \sqrt{\frac{A_2}{\pi}} \tag{8}$$

$$\tan\phi = \frac{dr}{dx} \quad ; \quad \tan\phi = \left(\frac{R_1}{2.l}\right)\frac{f_1(\alpha,\beta,K) + 2.f_2(\alpha,\beta,K).\bar{x}}{\sqrt{A_x}} \tag{9}$$

The detailed test conditions at different flow rates and another properties was shown in Table 1, The test duct has a rectangular section and test section entry cross-section was $A_1 = a.b = 0.028 m^2$ and exit cross-section area, $A_2 = c.d = 0.056 m^2$,

$$L_1 = 500, \; L_2 = 750, L_3 = 1000 \text{ and}$$
$$\alpha_{v,L_1} = 4^0, \alpha_{v,L_2} = 3^0, \alpha_{v,L_3} = 2^0 \text{ and, } \alpha_{a,L_1} = 8^0,$$
$$\alpha_{a,L_2} = 6^0, \alpha_{a,L_3} = 5^0, \; \phi_{L_1} = 6^0, \phi_{L_2} = 5^0, \phi_{L_3} = 4^0 \tag{10}$$

ϕ, symbols represent equivalence conical angle, α_v is angle between the top and bottom axes, α_a and is angle between the side edges. To characterize the in transition duct flow, mean velocity and local velocity profile measurement was carried out in a wide range of different $L_{1,2,3}$, $\alpha_{v,a}$, $\phi_{v,a}$ and flow rate. The distribution of the mean velocities and local in the velocities were measured by hot-wire anemometers at the five sections, $x/l = 0.20, 0.40, 0.60, 0.80, 1.00$ and in the each one section $[\, y/(b/2), z/(a/2)]$, $0 \le y/(b/2) \le 1 \;\; and \; 0 \le z/(a/2) \le 1$ for Reynolds number $Re = 2.10^5 - 6.10^5$, in which shown in Fig 1, There the Y – axis is in the horizontal direction and Z – axis is vertical directions. Prior to the experiments, we confirmed that flow distributions in all measuring sections were symmetric with respect to the symmetric axes of the cross-section, Y- and Z axes.

In this study, five measuring cross-section (as like Figure 1.b) was set up in the test duct, all the measurements were done 5 mm incremental displacement at the $x/l = 0.20, 0.40, 0.60, 0.80, 1.00$ downstream from the inlet test section.

The Reynolds number is defined as

$$Re = \frac{\rho \, Q.D_h}{\mu} = \frac{\dot{m} D_h}{A_{cross}\, \mu} \tag{11}$$

The pressure drop is defined by

$$\Delta P = f \, \frac{L}{D_h} \frac{\rho U_b^2}{2} \tag{12}$$

Here f is Darcy friction factor. The calculated friction coefficient was compared with Petukhov friction factor, [11], which is valid for $4000 \le Re \le 5.10^6$.

Table 1. Experimental conditions for pressure and velocity profile measurements

Flow rate $m^3/s x 10^{-3}$	$Re\, x 10^{-5}$	Measurement point (x/L)	$L_{1,2,3}\,(mm)$	ϕ^0	α_v^0	α_a^0	$A_1 (m^2)$	$A_2 (m^2)$
0.81	2	0.0, 0.2, 0.4,0.6,0.8,1.0	500	6	4	8	0.028	0.056
1.15	3		750	5	3	6		
1.25	4		1000	4	2	5		
1.56	5							
1.67	6							

3. Results and discussion

In transition pipe which is taken in hand, velocity measurements were taken for five sections in four various Reynolds number and three different pipe lengths. Measurements were taken both horizontal and vertical axes because there were no symmetry the parts in vertical and horizontal axes sides. Horizontal and vertical velocity profiles shown for each section were given for different pipe lengths in fig. 2...7. Symmetry local velocity profiles were given at the each $\bar{x}/l$, and along $y/(b/2), z/(a/2)$ from pipe axis to boundary.

Because of cross sectional area decrease along axis, velocity is decreased. Because of boundary layer development close to boundary velocity is decreased. Angle which between side edges and axis changes because of pipe length variations. Medium and long pipes vertical and horizontal velocity profiles are the same, and at dimensionless sections velocity changes from axis to boundary almost the same values (fig. 4...7).

Small length pipe's vertical velocity profiles (fig. 2), different from the medium and long ones but horizontal velocity profiles (fig. 3) were different each other for last tree sections. And velocity profiles in same sections at various Reynolds numbers were much more different. Using velocity profiles this last three sections it can be say that at small Reynolds number cross flow occurs. But increasing Reynolds number corrects regions close to boundary.

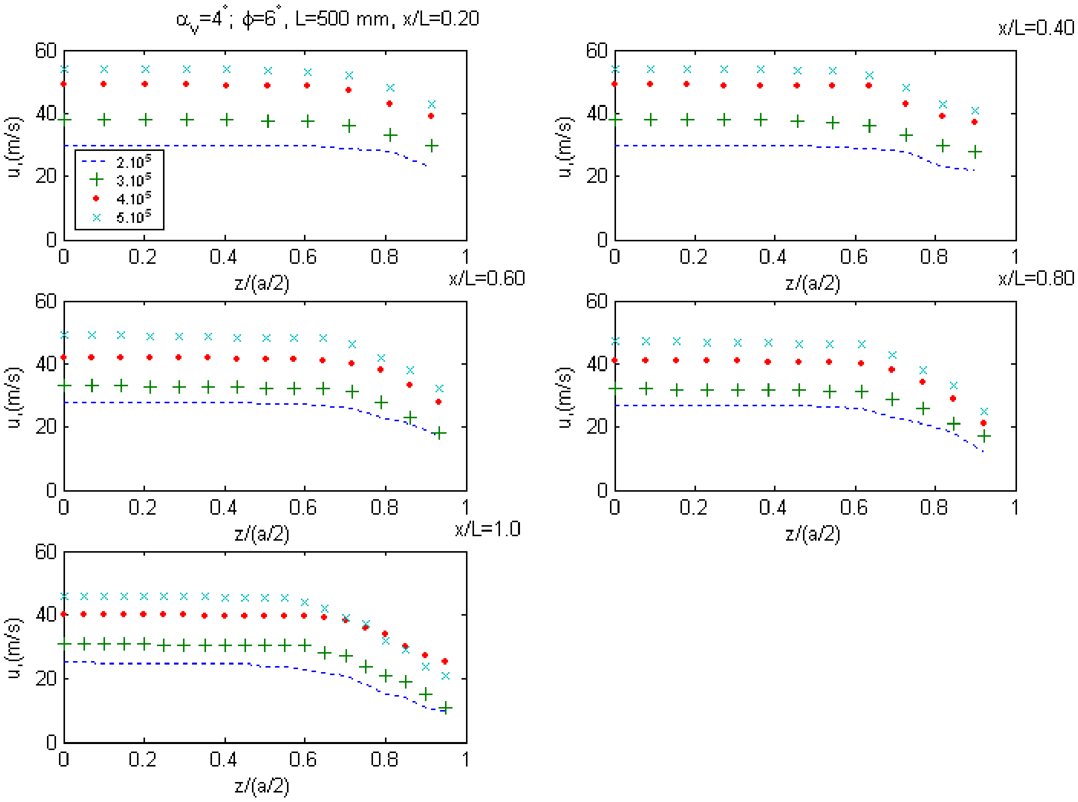

Fig. 2 – Measured V - velocity distributions $at\, the\, x/L = from 0.20\, to\, 1.00$ and $L = 500, \alpha_v = 4^0, \phi = 6^0$

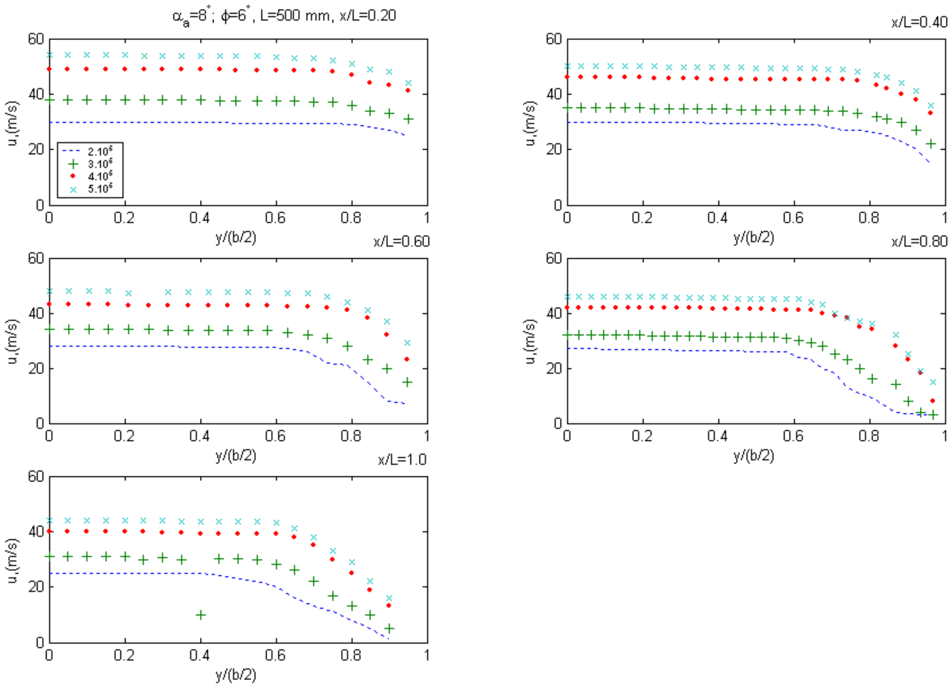

Fig. 3 – Measured U - velocity distributions $at\ the\ x/L = from 0.20\ to\ 1.00$ and $L = 500, \alpha_a = 8^0, \phi = 6^0$

This behaviour at horizontal axis arises from big angle between side edges and axis and small pipe length. Friction coefficient variations with Reynolds number for three pipe type are given in figure 8. The results show that when Reynolds number increase, friction loss coefficient decrease.

Biggest load loss coefficient was seen for smallest pipe and smallest load loss coefficient was seen for longest pipe. All these behaviours can be explained with flow corrections by means of increasing pipe length and pressure loss effects. Load losses for vertical axis bigger than horizontal axis for these transition parts. This behaviour arises from the bigger angle of side expansion angle and bigger horizontal boundary static pressure than vertical one. Static pressure increases because of transition part expansion throughout pipe length.

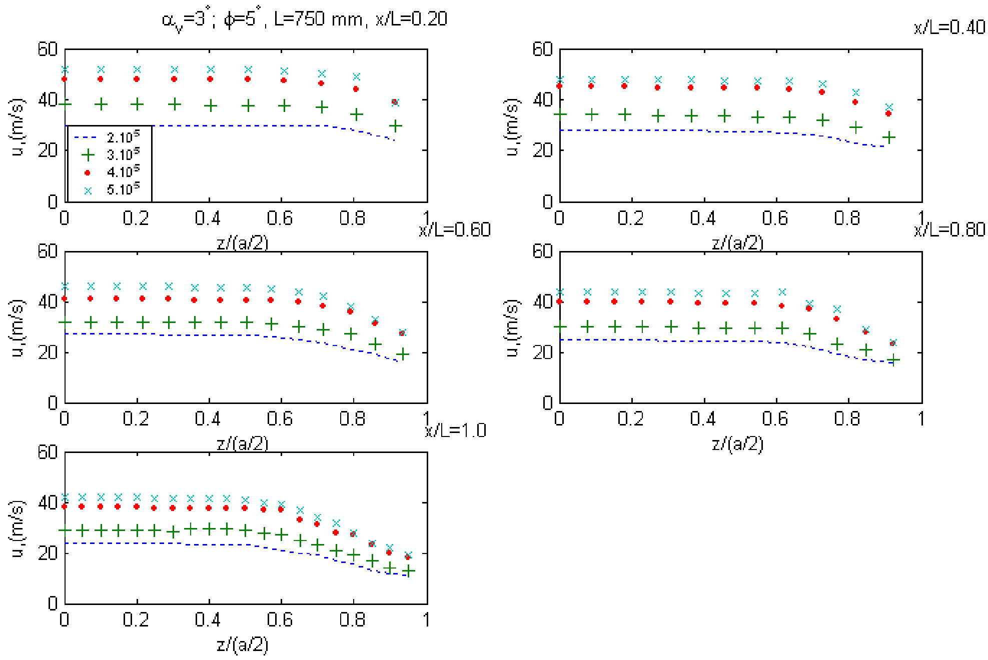

Fig. 4 – Measured V - velocity distributions $at\ the\ x/L = from 0.20\ to\ 1.00$ and $L = 750, \alpha_v = 3^0, \phi = 5^0$

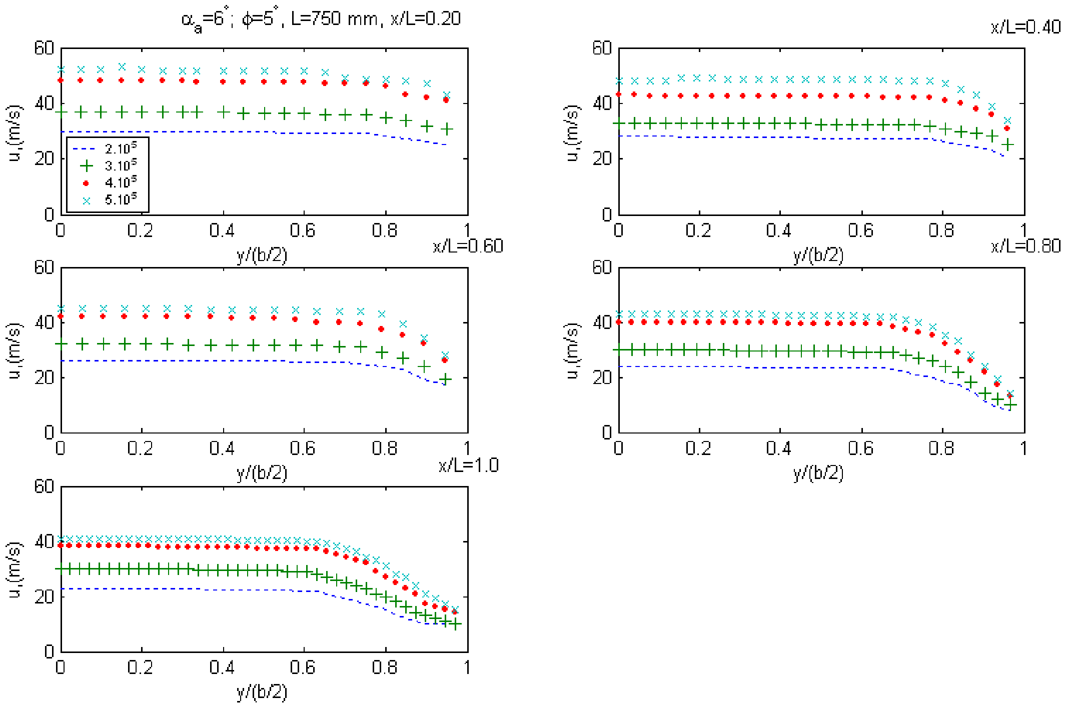

Fig. 5 – Measured U - velocity distributions $at\, the\, x/L = from 0.20\, to 1.00$ and $L = 750, \alpha_a = 6^0, \phi = 5^0$

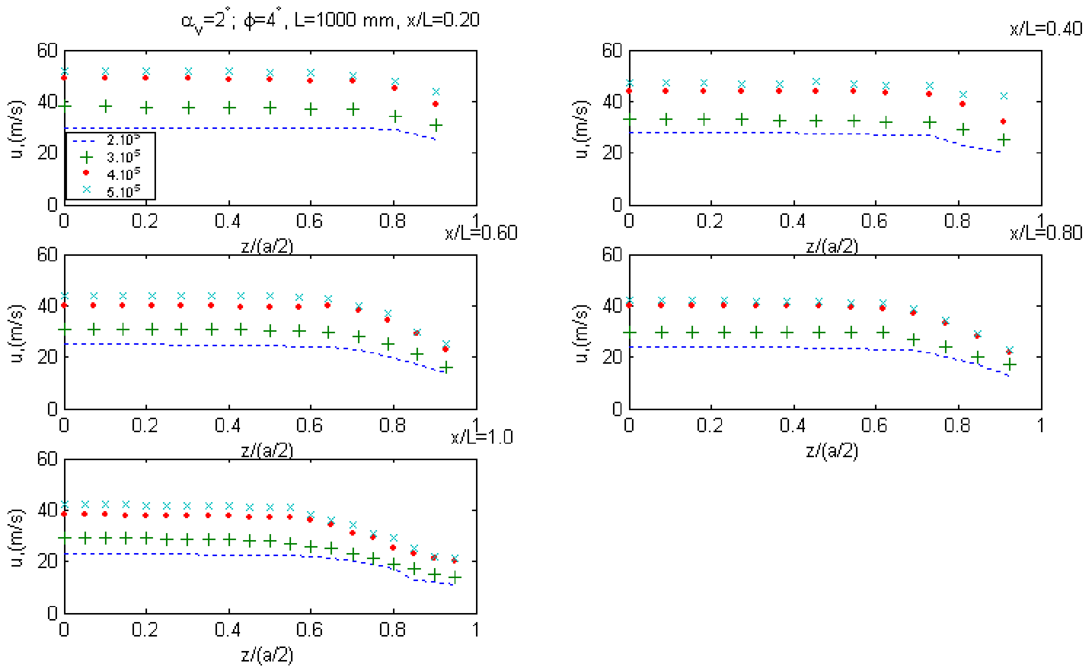

Fig. 6 – Measured V -velocity distributions $at\, the\, x/L = from 0.20\, to 1.00$ and $L = 1000, \alpha_v = 2^0, \phi = 4^0$

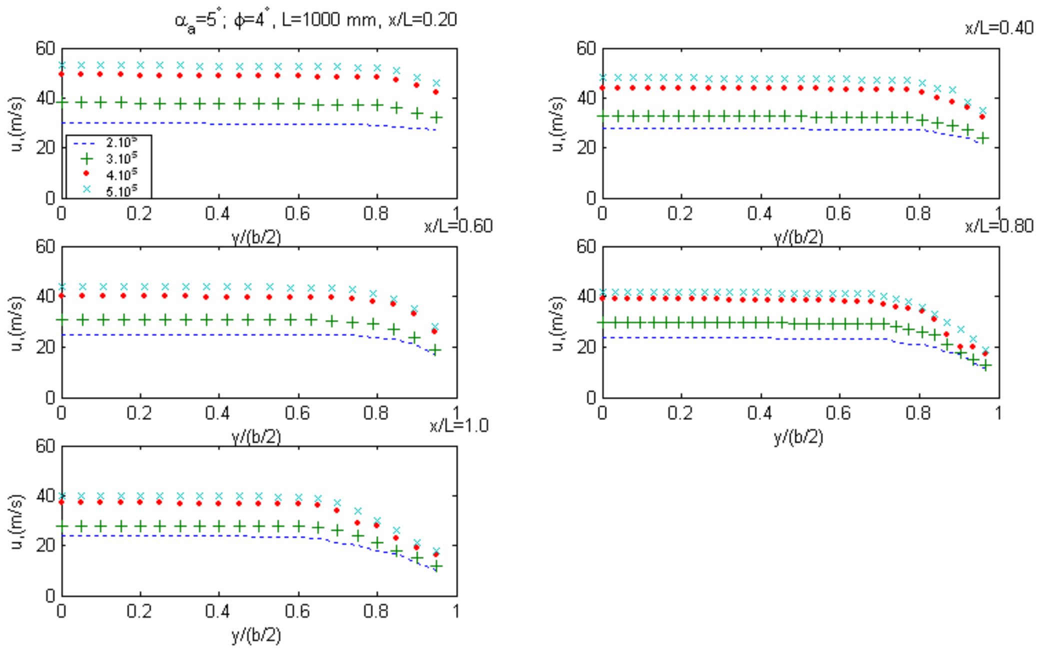

Fig. 7 – Measured U - velocity distributions $at\, the\, x/L = from 0.20\, to\, 1.00$ and $L = 1000, \alpha_a = 5^0, \phi = 4^0$

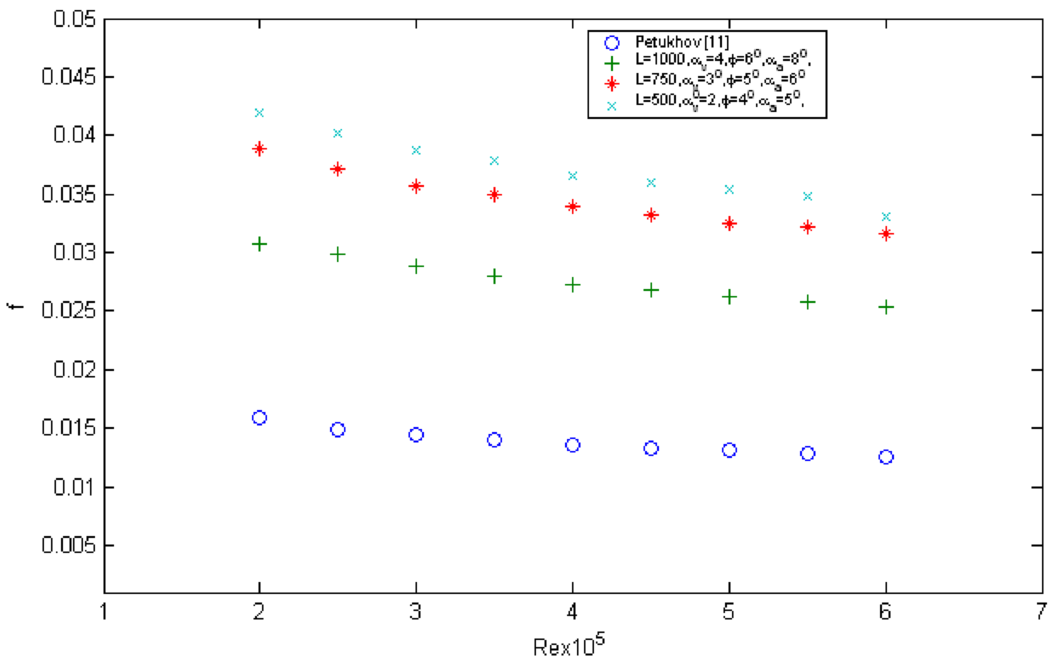

Fig. 8 – Friction factor distribution for various Re numbers

4. Conclusions

Present research work allows us to conclude:

- flowing velocity profile and loss coefficient results are obtained from the experiments done for three various lengths in rectangular to rectangular transition pipes;
- cross flows occurs in horizontal side for small length transition pipes; this behaviour is the results of the big angle value between side edges with axis;
- the biggest load loss coefficient value was seen in small pipes;
- load loss coefficient decreases with increasing pipe length and Reynolds number;
- when the angle is small between pipe axis and pipe surface, load loss coefficient is small with respect to big angle ride; this behaviour is the result of angular expansion.

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